AN EJECTOR FOR ACCELERATING COAL PARTICLES

A THESIS

Presented to

the Faculty of the Division of Graduate Studies

Georgia Institute of Technology

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Mechanical Engineering

by

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December 1950

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COAL PARTICLES

Approved:

Date Approved by Chairman December 12, 1950

ACKNOWLEDGMENTS

The writer is indebted to Professor M. J. Goglia of the School of Mechanical Engineering who suggested this problem. He assigned valuable literature and offered me continual and helpful advice.

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A.......Cross-sectional area, sq. in. or sq. ft.

c_p......Specific heat at constant pressure, for air c_p = 0.241 B.T.U./lb.-°F.

D.......Diameter of mixing tube, inches.

d.......Diameter of metering element, inches.

g.......Gravitational acceleration, taken as the standard value, 32.2 ft./sec.^{2}

h.......Enthalpy per unit mass, B.T.U./lb.

J.......Mechanical equivalent of heat, 778 ft.-lb./B.T.U.

K.......Ratio of specific heat at constant pressure to specific heat at constant volume, for air K = 1.400

K.......Flow coefficient, coefficient of discharge with approach factor included.

L.......Length of mixing tube, inches.

$\bar{m}$.......Molecular weight, for air $\bar{m}$ = 28.97

n.......Polytropic exponent.

p.......Pressure, lbs./in.^{2}, or lbs./ft.^{2}

Δp......Differential pressure across metering element, lbs./in.^{2}

$\bar{R}$.......Universal gas constant, 1545.32 ft. lb./lb.-mole-°F.

R.......Gas constant = $\bar{R}/\bar{m}$, for air R = 53.3 lb.-ft./lb.-°F

R.......Radius of mixing tube, inches.

R.......Reynolds number = $48W/\pi d \mu$

r.......Pressure ratio.

T.......Absolute temperature, °F. abs.

t.......Fahrenheit temperature, °F.

V.......Velocity, feet per second.

v.......Specific volume, ft.^{3}/lb.

W.......Mass rate of flow, lbs./sec.

X.......Radius of traverse, inches.

Y.......Empirical expansion factor for fluid, dimensionless.

μ.......Absolute viscosity, lbs./sec.-ft.

ρ.......Density, $lbs./ft.^3$

Subscripts

1.......Refers to section 1, figure 2.

2.......Refers to section 2, figure 2.

i.......Refers to initial conditions of primary fluid.

o.......Refers to initial conditions of secondary fluid; Equations 1 and 2 refers to isentropic stagnation conditions.

t.......Refers to conditions at the throat of primary nozzle.

Superscripts

'.......Refers to primary flow.

".......Refers to secondary flow.

AN EJECTOR FOR ACCELERATING

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SUMMARY

A preliminary design and experimental investigation was carried out to determine the accelerating characteristics of an annular type ejector. Air was used for both the primary and secondary fluids.

Both total and static pressures were recorded at the exit of five constant area mixing tubes ranging in length from an L/D ratio of two to ten. Static pressures were negligible and disregarded in velocity calculations.

Total pressure profiles indicated that a uniform velocity distribution was attained in a relatively short mixing tube, an L/D of approximately ten. Shorter mixing tubes indicated only partial mixing; the shortest tube indicated no mixing at exit.

Experimental evidence indicates unsymmetrical total pressure profiles at exit of each mixing tube for an L/D of four to eight. Such results are believed to indicate a characteristic of the mixing process.

Included in this investigation are entrainment characteristics indicating the proper mixing tube length,

an L/D of six, for maximum secondary to primary flow. This result corroborates data published in earlier investigations.

The scope of this preliminary investigation did not include the analysis which describes the interaction between a supersonic stream and a subsonic stream.

It is expected that the performance of this ejector when used to accelerate particles will yield information concerning the mechanism of momentum transfer from the high velocity annular primary air jet to the central particle laden secondary induced air stream.

INTRODUCTION

Simple and inexpensive methods of pulverizing coal particles for use as a fuel in a coal burning gas turbine have recently been investigated by the Locomotive Development Committee (16)* of Bituminous Coal Research, Inc. The present method requires bulky and expensive machinery (17).

An experimental method of pulverization suggest bombarding a sufficiently hard target with coal particles. Experimental equipment, consisting chiefly of a high pressure air nozzle, arranged for the introduction of coal particles has proven satisfactory. However, this design requires excessive high pressure air for operation and is relatively expensive for practical use. This undesirable feature led to the investigation of a more practical method.

The object of this thesis was to make a preliminary study of an ejector and to design a test model for accelerating coal particles. These particles, of the order of 100 microns, are to be accelerated from rest to a predetermined terminal velocity of 600 feet per second (16) which is sufficient for coal pulverization. Such a design, as compared to existing apparatus, would be less complicated and require a minimum of high pressure air for operation.

The literature concerning ejectors reveals a well

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* Numbers in parenthesis refer to Bibliography.

developed theory with good test correlation for central type ejectors, Fig. 1.

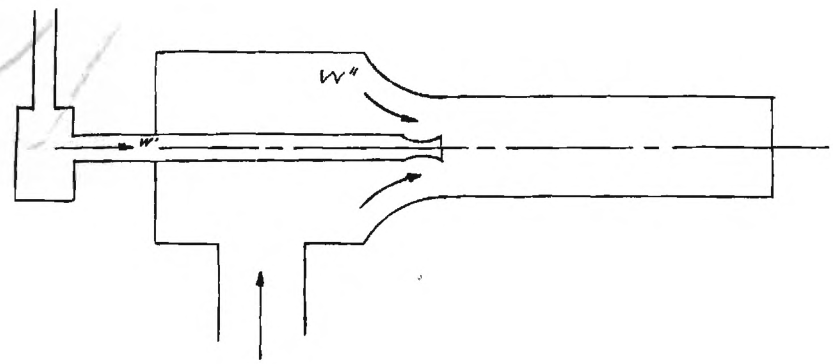

Fig. 1 Central Type Ejector

Keenan and Neumann (8) applied a one-dimensional analysis to a central type air ejector. This analysis covered the two cases, (a) mixing at constant pressure, and (b) mixing at constant area. Test data were obtained for constant area mixing using large ratios of mixing tube area to primary nozzle area and relatively small primary air pressures. A later paper by Keenan, Neumann, and Lustwerk (9) greatly extended the range of variables. Assumptions other than constant pressure or constant area mixing yielded no satisfactory analysis.

Recently, an investigation to determine the performance of annular type ejectors, Fig. 2, was conducted at Stanford University under the sponsorship of the National Advisory Committee for Aeronautics. Figure 2 shows a convergent-divergent nozzle in the primary stream. Only a Convergent nozzle was used for the N.A.C.A. tests.

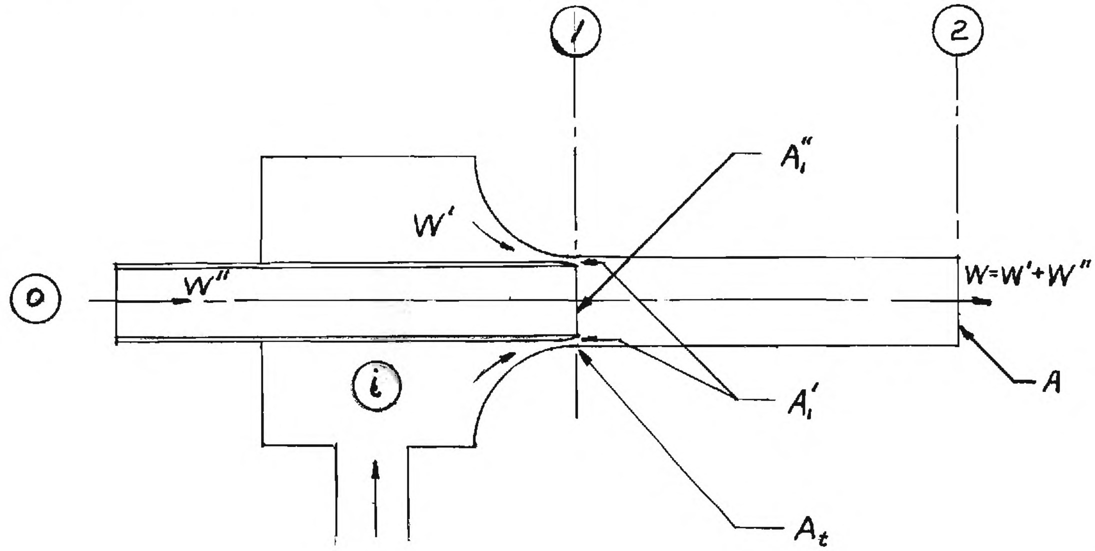

Fig. 2 Annular Type Ejector

The performance of annular type ejectors with constant area mixing tubes indicated terminal velocity and entrainment characteristics equivalent to those of the central type. A combination of annular jet and divergent mixing tubes resulted in an improvement in the entrainment characteristics, however, this combination exhibited a very poor velocity distribution at exit.

Much time will be required to acomplish the final objective; here, it was felt that a study of the design, including preliminary air tests, would be sufficient. Future tests with the introduction of coal particles are planned on the basis of the present investigation.

APPARATUS

The general arrangement of the experimental apparatus is shown in figure 3. Figures 4, and 5 view close-up details of the model. The arrangements for measuring total and static pressures at exit of each mixing tube, and the mercury manometers for reading differential pressures are also shown in detail. Temperatures were measured with mercury in glass thermometers. A pressure gage attached to the model housing indicated plenum chamber pressures. A pressure regulator, installed in the primary air line, minimized pressure fluctuations.

Primary air was supplied by two reciprocating compressors, one large three cylinder laboratory unit, and a smaller one cylinder compressor connected in parallel. This arrangement was made necessary so as to maintain a continuous flow of high pressure primary air.

The flow of primary air was metered by using a thin plate orifice, 0.754 inches diameter. The orifice plate was located concentrically between two pipe flanges in a two inch pipe line. Flange taps, drilled one inch upstream and downstream from the orifice face, were used to measure differential pressures across the orifice. The thin plate orifice, and the location and machining of the pressure taps were made in accordance with specifications outlined in the American Society of Mechanical Engineers Test Code(2).

Flow coefficients are given in terms of Reynolds number and ratio of orifice to pipe diameter. Such coefficients were selected as the average from a series of tests and fall within a tolerance of ± 0.5 percent.

Secondary air was metered by the nozzle shown in detail in figure 10. A pressure tap, in the straight section of the nozzle passage, was connected to one leg of an open end mercury manometer, thereby, indicating differential pressures across the nozzle, and the static pressure at section 1, (Fig. 2), neglecting a small pressure drop due to friction. A coefficient for this nozzle was assumed in accordance with reference 12.

Both total and static pressures were recorded at the exit of each mixing tube. Total pressures were indicated by a mercury manometer; static pressures were indicated by a small water manometer. Two separate hypodermic needles were used to indicate these pressures. The static pressure needle was soldered closed at the upstream end, and a small pressure tap drilled in one side about one-eight inch from this end. The downstream end of each hypodermic needle was soldered to a small copper tubing bent perpendicular to the direction of flow so as to avoid any obstruction to the discharge stream. The copper tubing was clamped to a screw-type crosshead with a scale attached to the movable portion. Traverse points across the mixing tube diameter could be located within one sixtyfourth of an inch.

With the total pressure needle clamped in position a traverse was made and recorded. After the traverse was completed, the total pressure needle was removed and the static pressure needle clamped in the same position. With the static pressure tap placed in the exit of the mixing tube, a second traverse was made. Both total and static pressure measurements were observed under the same conditions for flow through each mixing tube.

The correct radii for the primary and secondary air nozzles were machined to size by first grinding a cutting tool to the correct dimensions. The spacing at the throat of the primary nozzle was checked with shims after assembly of the model. The detail of the secondary nozzle is shown in Fig. 10, and the assembly drawing, Fig. 7, shows the primary nozzle.

Plastic mixing tubes were inserted with the future intention of viewing the mixing process. From the working drawings, it can be seen that the exit of the nozzle was machined for a press fit with each mixing tube. A section of the nozzle rim was milled away for the added convenience of viewing the mixing process.

DISCUSSION

Design Considerations

The simplicity of transporting suspended particles in a straight duct led to the choice of employing an annular type ejector; such a unit permits acceleration of the central particle laden secondary air stream by means of the annular primary jet. With the central type ejector, where sharp turns are encountered in the secondary air passage, the problem of conveying suspended particles is more complicated.

Two methods of mixing have been studied (8), constant pressure mixing, and constant area mixing. Ejectors have been analyzed on the basis of one-dimensional flow, and two-dimensional flow; so far a two-dimensional analysis (9) has not proven satisfactory for design purposes. A constant area mixing tube design was used here to facilitate analysis of experimental results of momentum transfer between the primary stream and the particle laden secondary stream.

Preliminary calculations for this problem indicated that either a small diameter of mixing tube, or an expansion of the primary stream to supersonic velocity would be required to achieve the prescribed terminal velocity of 600 feet per second (16). The supersonic primary stream appeared to be the more practical choice.

Referring to figure 2, the two streams are exposed for mixing at the exit of the primary nozzle. During the mixing process a transfer of momentum from the high speed annular jet to the slower moving central jet accelerates the secondary stream providing a uniform velocity profile at exit of the mixing tube. The resulting stream discharges to the atmosphere. For coal pulverization a complete arrangement would require an apparatus for admitting the coal particles into the secondary air together with a suitable target placed at exit of the mixing tube. The fluidized stream, consisting of air and coal particles, would then be accelerated from rest to the prescribed terminal velocity; coal particles would bombard the target, thereby permitting some pulverization.

One-dimensional considerations do not provide a means for calculating the proper mixing tube length for achieving uniform terminal velocity. Experimental tests were resorted to for the determination of the proper ratio of L/D for the constant area mixing tube for complete mixing.

Prescribed conditions for this problem were 60 p.s.i. for plenum chamber pressure, section i, and 14.7 p.s.i. at 80 °F for the secondary stream, section o.

Method od Analysis

With reference to figure 2, the assumption is made that both streams are initially at rest and both fluids expand reversibly and adiabatically to section 1. From the equations of flow, the velocity at section 1, for both streams, can be calculated from the relation (10).

$$V_1 = \sqrt{\frac{T_0}{\bar{m}}} \sqrt{\frac{2g\bar{R}n}{n-1}} \sqrt{1-r^{\frac{n-1}{n}}} \qquad 1$$

The mass rate of flow, at section 1, for each stream, can be calculated from the relation (10).

$$W_1 = AP_0 \sqrt{\frac{\bar{m}}{T_0}} \sqrt{\frac{2g(n)}{\bar{R}(n-1)}}\, r^{\frac{1}{n}} \sqrt{1-r^{\frac{n-1}{n}}} \qquad 2$$

Further the assumption is made that both streams mix adiabatically and discharge from the exit of the mixing tube with a uniform velocity profile. Both streams are assumed to have the same molecular weight and ratio of specific heats.

Between sections i and o, where the two streams are considered at rest, to section 2, where mixing is complete use is made of the relations involving conservation of mass, energy, and momentum, viz

The Continuity equation

$$W = W' + W'' \qquad 3$$

The Energy equation

$$W'h_i + W''h_0 = W\left(h_2 + \frac{V_2^2}{2g}\right) \qquad 4$$

The Momentum equation

$$\frac{W'V_1'}{g} + \frac{W''V_1''}{g} + P_1 A = \frac{WV_2}{g} + P_2 A \qquad 5$$

In addition the enthalpy

$$h = c_p T \qquad 6$$

and the continuity equation in the form

$$V = \frac{Wv}{A} \qquad 7$$

are employed. Further the initial temperatures of the primary and secondary fluids are assumed equal, $t_i = t_o$, giving $h_i = h_o$. From equations 4 and 6 the energy equation becomes

$$c_p T_i = \frac{K}{K-1} P_2 v_2 + \left(\frac{W}{A}\right)^2 \frac{v_2^2}{2g} \qquad 8$$

or solving for v_2

$$v_2 = \frac{-P_2 \frac{K}{K-1} \pm \sqrt{\left(P_2 \frac{K}{K-1}\right)^2 + 4\left[\frac{1}{2g}\left(\frac{W}{A}\right)^2 (c_p T_i)\right]}}{\frac{1}{g}\left(\frac{W}{A}\right)^2} \qquad 9$$

Thus all the relations for this analysis are given.

<u>Method of Solution</u>

Because of the form of the above equations, a solution by successive approximations must be employed.

First, select a size for the constant area mixing tube, and assume an area ratio of primary nozzle exit area to the mixing tube area. With a given or chosen value for the primary pressure, section i, assume a value for the pressure at section 1. With the aid of equations 1 and 2, the primary and secondary fluid velocities V_1' and V_1'', and the mass rates of flow W' and W'' can be calculated. From equation 9 the specific volume at section 2 can now be found. Substituting in equation 7 gives the average velocity at exit of the mixing tube. If this value of V_2 is too high or too low, new values for the areas and/ or primary pressure should be selected until the required velocity is obtained. From the above calculations, all the quantities for substitution in the momentum equation are available. If this equation is not satisfied, a new value of the pressure at section 1 should be assumed and the above procedure repeated.

It is pointed out that this test model was designed to operate at 60 p.s.i.g., but unfortunately had to be operated at 55 p.s.i.g. because of present laboratory facilities. All theoretical calculations were made on the basis of operating at 60 p.s.i.g.

Mixing Tube Efflux Characteristics

The actual velocity was calculated from total pressure readings using the relation (1)

$$V_{avg.} = \sqrt{2g}\left[\sqrt{P/\rho}\,\right]_{avg.}$$

Total pressure readings were taken at the mean radius representing five equal areas of flow. Ten readings were recorded across the diameter at distances from either side of the center:

$$X = \sqrt{.1}\,R,\ \sqrt{.3}\,R,\ \sqrt{.5}\,R,\ \sqrt{.7}\,R,\ \sqrt{.9}\,R.$$

In the ideal case, static pressures at exit of each mixing tube are assumed to be atmospheric, however, both postive and negative readings were recorded. These pressures amounted to only a few inches of water, and were not included in velocity computations. The average velocity at the mixing tube exit, calculated from experimental data, was 660 feet per second. The theoretical velocity at exit amounted to 753 feet per second.

Entrainment Characteristics

The actual mass rates of flow for both primary and secondary air were calculated from the relation (2)

$$W = .668\,AKY\sqrt{e\Delta P}$$

The flow coefficient for the orifice, 0.605, was taken from reference 1. Reynolds number for the primary fluid was approximately 95,000. The flow coefficient for the nozzle, 0.98, was assumed in accordance with reference 12. Reynolds number for the secondary fluid was in the neighborhood of 150,000. The empirical expansion factor, Y, was taken to be 1.00 for flow through the orifice and nozzle. Actual mass rates of flow for secondary and primary air are given in Table I.

Table I Actual Mass Rates of Flow

L/D	2	4	6	8	10
W''	0.0564	0.0745	0.0815	0.0798	0.0766
W'	0.0612	0.0589	0.0589	0.0590	0.0567
W''/W'	0.921	1.265	1.385	1.352	1.351

Theoretical calculations for the mass flow of secondary and primary air gave 0.096, and 0.082 pounds per second respectively. Experimental data corroborates earlier investigations (8) indicating a mixing tube length of L/D of approximately 6 for maximum primary to secondary mass flow.

Primary Nozzle Characteristics.

Theoretical calculations gave the velocity at exit of the primary nozzle to be 1640 feet per second; the throat velocity was calculated to be 1035 feet per second. From experimental data the maximum primary jet velocity issuing from the minimum length of mixing tube, ($1\frac{1}{2}$ inches), was calculated to be 1160 feet per second. While some over-expansion, due to operating at a slightly reduced pressure, may account for a part of this discrepancy, losses are due chiefly to a compression shock.

It was initially appreciated that the discharge characteristics of the primary nozzle would substantially influence the results. The continuity and energy equations are written between sections 1,o and 2, while the momentum equation is written between sections 1 and 2. A more complete analysis must take into account the presence of a shock.

<u>A Discussion of Momentum Transfer</u>

Figure 6 shows graphically total pressures at exit of each mixing tube. Neglecting static pressure variations, velocity profiles would be represented by similar curves. Since the traverse represents equal mass rates of flow, these curves may be interpreted to indicate the progressive transfer of momentum between the two streams. It is seen that mixing took place throughout the region for an L/D of approximately 4 to 8. Transfer of momentum in this region is not symmetrical across a horizontal diameter. The reason for this result is not immediately clear, however, it is believed due to a characteristic of the mixing process, indicating helical swirls or vortices. Results indicate that the mixing process was complete for an L/D of approximately 10.

Tables II through VI list the discharge survey data for each mixing tube; experimental data necessary for calculating the primary and secondary flow is also included.

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

Preliminary tests for this design, operating under prescribed conditions, indicate that a uniform velocity profile can be obtained in a constant area mixing tube length of L/D of approximately 10. The proper length of mixing tube for maximum flow of secondary to primary fluid, $W''/W' = 1.385$, was an L/D of 6.

Therefore for this problem an L/D of approximately 10 would provide the required terminal velocity. This mixing tube length corresponds to a mass flow ratio of 1.351 of secondary to primary fluid.

This design was to operate with the added stipulation of requiring a minimum of primary air. The ejector analysis shows that for initial conditions of the two fluids at rest, the terminal velocity is a function of the mass flow , area of mixing tube, and specific volume at exit. The mass flow and specific volume obviously depend on the selected values of primary and secondary nozzle areas. For a mixing tube diameter of 3/4 inch, an area ratio of 6 was required to achieve the prescribed terminal velocity. The amount of primary air was fixed, Table I.

The results of this thesis indicate further study is needed (a) to investigate the effects of a compression shock in the primary air stream, and (b) to verify unsymetrical velocity distributions during mixing.

Calculations for the designed ejector operating at 55 p.s.i.g. indicate that the theoretical secondary and primary mass rates of air flow are 0.0978 and 0.0795 pounds per second respectively. The terminal velocity for operation at 55 p.s.i.g. amounts to 740 feet per second. This theoretical velocity differs from the 660 feet per second which was measured experimentally. The possible reasons for this apparent difference are that the analysis assumed first the absence of any discontinuities such as shocks and second a frictionless mixing tube was assumed.

Recommendations

A well designed rake with permanently fixed impact tubes to fit over the exit of the mixing tube would eliminate experimental errors in velocity measurements.

The performance of this ejector should be observed more completely using first air alone and second suspended particles in the secondary stream. The results of the latter observations should be used to modify the design procedure for ejectors used to accelerate particles in suspension.

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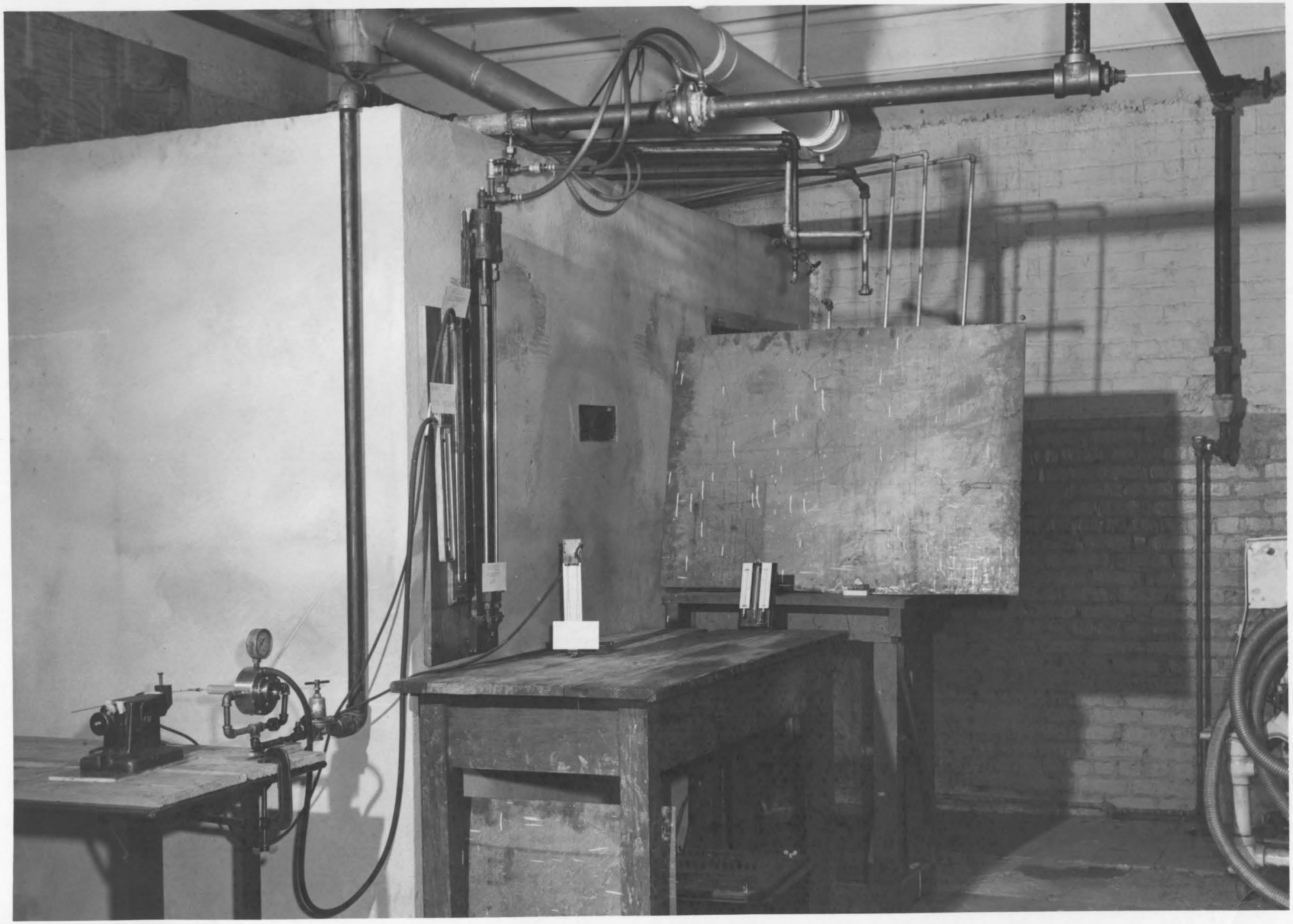


FIG. 3, GENERAL ARRANGEMENT OF APPARATUS

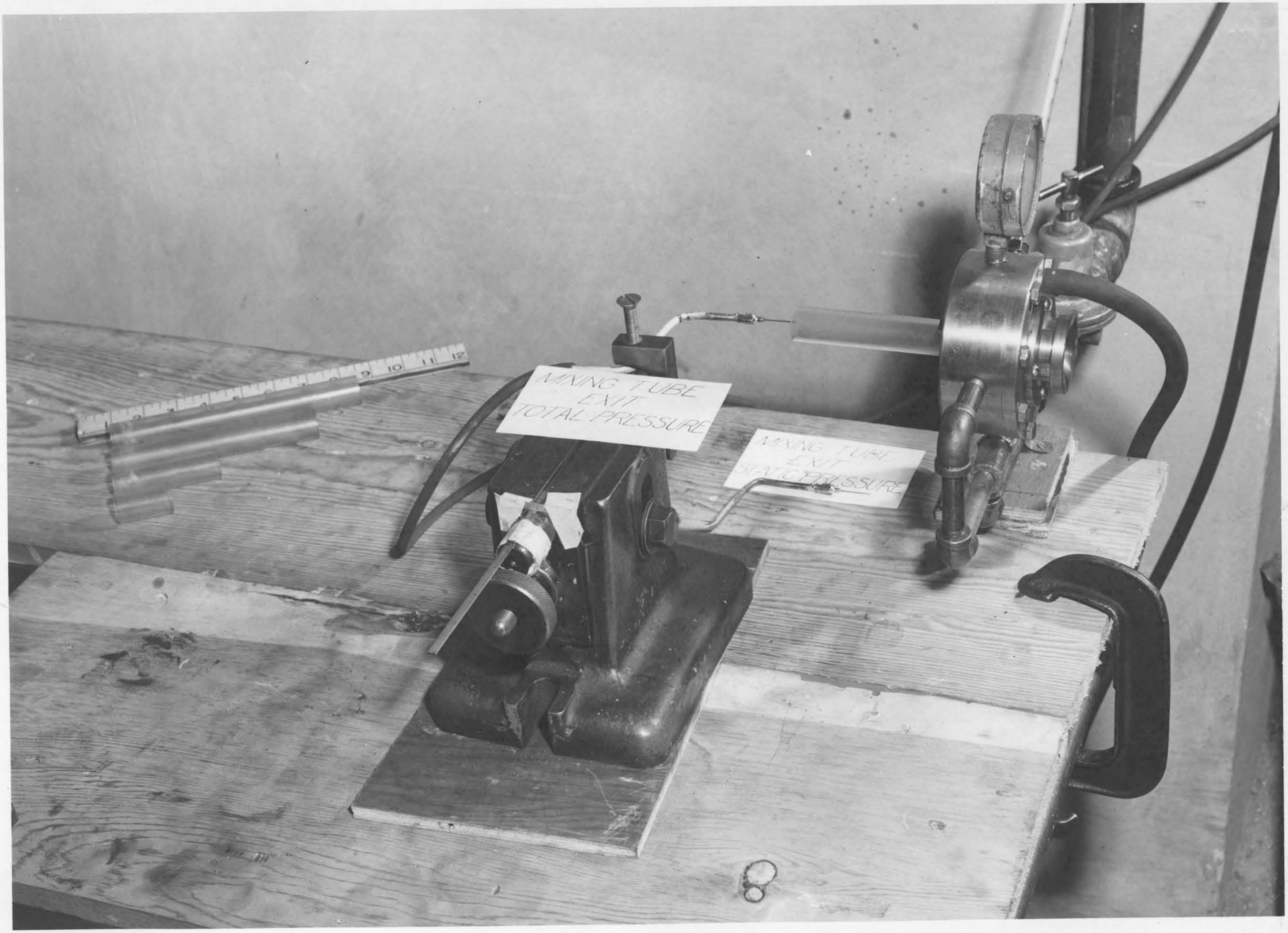


FIG 4, DETAIL OF MODEL AND SURVEY APPARATUS

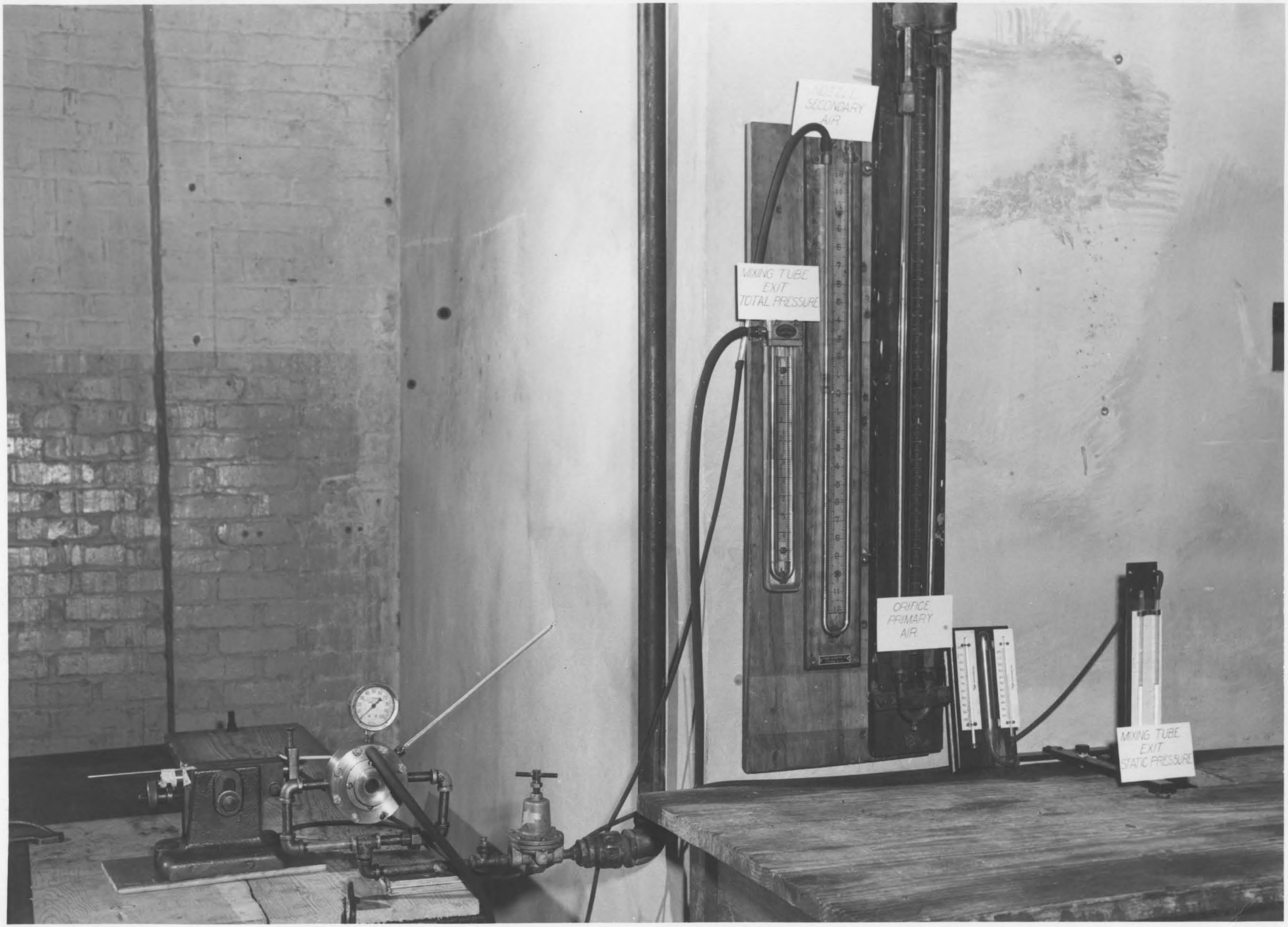


FIG. 5, MODEL AND INSTRUMENTATION

EXPERIMENTAL DATA

Barometric pressure ....28.98 in. Hg.
Pressure before orifice ....93.0 p.s.i.g.
Plenum chamber pressure ....55.0 p.s.i.g.
Ambient-air temperature ....82.0 °F.
Air temperature before orifice ....82.0 °F.
Plenum chamber temperature ....79.0 °F.

Table II, Discharge Survey Data, L/D = 2

Nozzle differential pressure (secondary air)..7/16 in. Hg.
Orifice differential pressure (primary air)...2.10 in. Hg.

Radius inch	Total pressure "Hg.	Static pressure "H_2O
23/64	21.0	+5.6
20/64	17.0	+1.4
17/64	8.4	+0.6
13/64	2.3	−0.2
8/64	0.3	0.0
center	0.0	−0.2
8/64	0.0	−0.2
13/64	0.1	−0.2
17/64	2.5	−0.1
20/64	10.0	+0.6
23/64	19.4	+3.0

Table III, Discharge Survey Data L/D = 4

Nozzle differential pressure (secondary air)..13/32 in. Hg.
Orifice differential pressure (primary air)...3.60 in. Hg.

Radius inch	Total pressure "Hg.	Static pressure "H_2O
23/64	12.6	+3.4
20/64	12.6	+4.2
17/64	11.5	+2.9
13/64	8.0	+2.1
8/64	7.0	+1.2
center	4.0	−1.0
8/64	1.5	−2.2
13/64	2.0	−1.0
17/64	5.0	−0.4
20/64	8.0	−0.2
23/64	11.0	+0.8

Table IV, Discharge Survey Data, L/D = 6

Nozzle differential pressure (secondary air)..13/32 in. Hg.
Orifice differential pressure (primary air)... 4.30 in. Hg.

Radius inch	Total pressure "Hg.	Static pressure "H_2O
23/64	8.2	+1.0
20/64	10.0	+1.0
17/64	10.0	+0.6
13/64	10.0	−0.2
8/64	10.8	−1.2
center	9.5	−2.3
8/64	5.5	−1.6
13/64	4.2	−1.0
17/64	5.0	−0.2
20/64	6.6	+0.2
23/64	6.8	+1.0

Table V, Discharge Survey Data, L/D = 8

Nozzle differential pressure (secondary air)..13/32 in. Hg.
Orifice differential pressure (primary air).. 4.10 in. Hg.

Radius inch	Total pressure "Hg.	Static pressure" H_2O
23/64	6.2	+1.1
20/64	8.3	+1.3
17/64	9.2	+1.3
13/64	9.5	+1.1
8/64	10.2	+0.5
center	9.2	−1.0
8/64	7.5	−1.4
13/64	6.2	−0.8
17/64	5.9	−0.4
20/64	5.9	−0.2
23/64	5.5	+0.3

Table VI, Discharge Survey Data, L/D = 10

Nozzle differential pressure (secondary air).. 3/8 in. Hg.
Orifice differential pressure (primary air)... 3.80 in. Hg.

Radius inch	Total pressure "Hg.	Static pressure "H_2O
23/64	5.5	+0.8
20/64	6.8	+1.6
17/64	7.6	+2.0
13/64	8.6	+2.2
8/64	9.2	+2.2
center	9.0	+1.8
8/64	8.0	+1.6
13/64	7.0	+1.7
17/64	6.2	+1.8
20/64	5.8	+1.9
23/64	4.6	+1.5

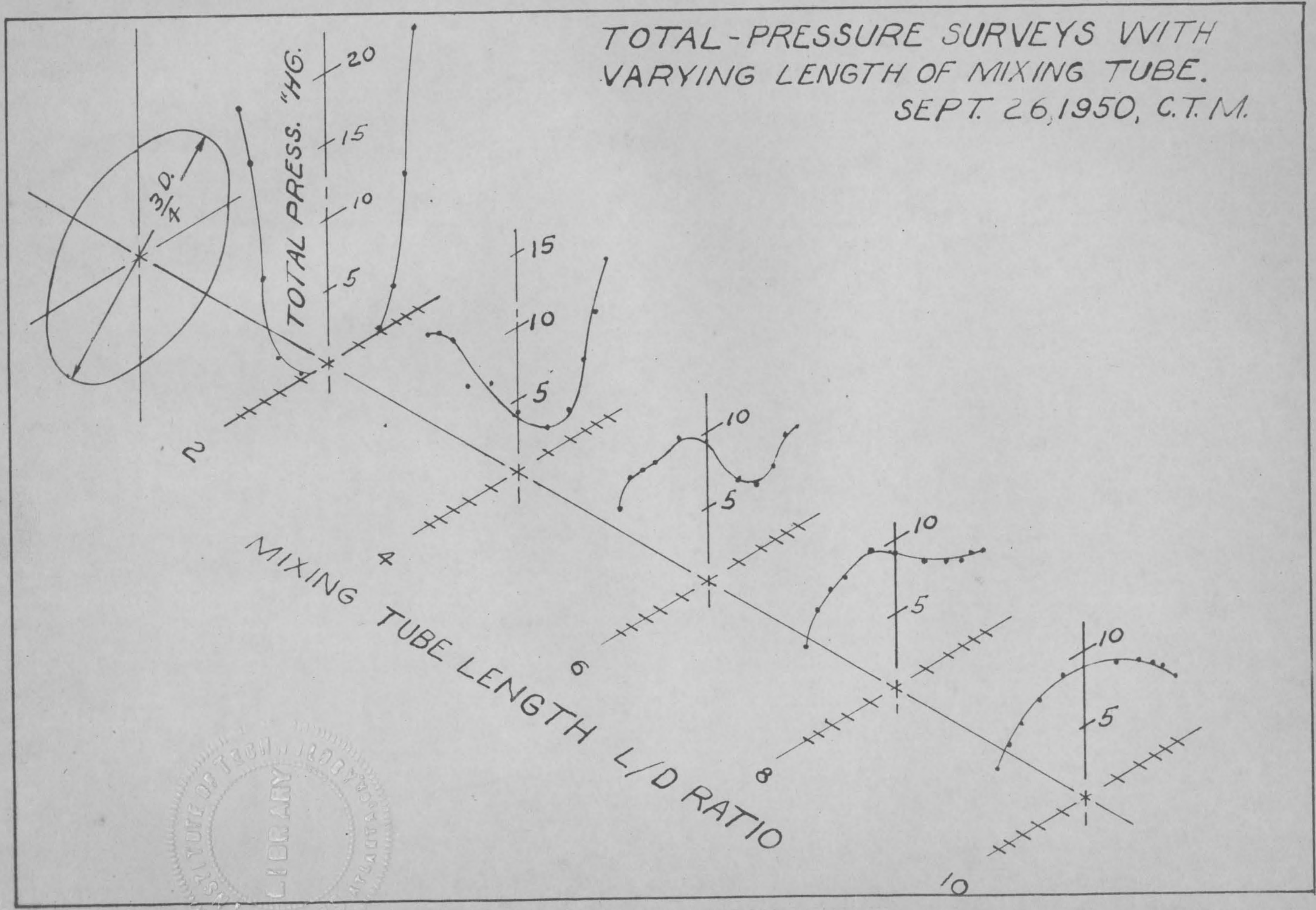


Figure 6, Total Pressure Surveys

APPENDIX

Design Calculations

For operation on air the initial conditions of pressure and temperature are given.

$p' = 60$ p.s.i.g. $\qquad t_i = 81\ ^{o}F.$

$p'' = 14.23$ p.s.i.g. $\qquad t_o = 81\ ^{o}F.$

Select a mixing tube size, D = 3/4 inch, giving a mixing tube area, A = 0.00307 sq. ft. Assume a ratio of mixing tube area to primary nozzle exit area, $A/A_1' = 6$.

$A_1' = 0.00307/6 = 0.000512$ sq. ft.

$A_1'' = 0.00307 - 0.000512 = 0.002558$ sq. ft.

Now assume a pressure at section 1, $p_1 = 11.52$ p.s.i.

$p_1/p'' = 0.810, \qquad p_1/p' = 0.810(14.23)/(74.23) = 0.155$

The velocity at section 1 may be calculated from equation 1.

$V_1' = 4.32(589.94)(0.64262) = 1640$ feet per sec.

$V_1'' = 4.32(589.94)(0.24171) = 615$ feet per sec.

The mass rate of flow may be calculated from equation 2.

$W_1' = 0.000512(74.23)(144)(0.38176)(0.16967)/4.32 = 0.082$ lbs./sec

$W_1'' = 0.002558(14.23)(144)(0.38176)(0.20795)/4.32 = 0.096$ lbs./sec

$W = W' + W'' = 0.178$ lbs./sec

The specific volume at exit may be found from equation 9.

$$V_2 = \frac{-14.23(144)\frac{1.4}{.4} \pm \sqrt{\left[\frac{14.23(144)(1.4)}{.4}\right]^2 + 4\left[\frac{1}{64.4}\left(\frac{0.178}{0.00307}\right)^2(.240)(778)(541)\right]}}{\frac{1}{32.2}\left(\frac{0.178}{0.00307}\right)^2}$$

$v_2 = 12.99$ cubic feet per second.

Solve for the velocity at exit from equation 7.

$$V_2 = (0.178)(12.99)/0.00307 = 753 \text{ feet per sec.}$$

From reference 13, it was estimated that the actual velocity at exit would amount to approximately 85% of the theoretical velocity or 640 feet per sec.

Now check to see that the above values satisfy the momentum equation.

$$(0.082)(1640)/32.2 + (0.096)(615)/32.2 + 11.52(144)(0.00307) = (0.178)(753)/32.2 + 14.23(144)(0.00307)$$

$$11.12 \neq 10.46$$

While a difference of 0.66 indicates a slight error in the assumption of the pressure at section 1, it falls within the accuracy of slide rule calculations.

Throat Area

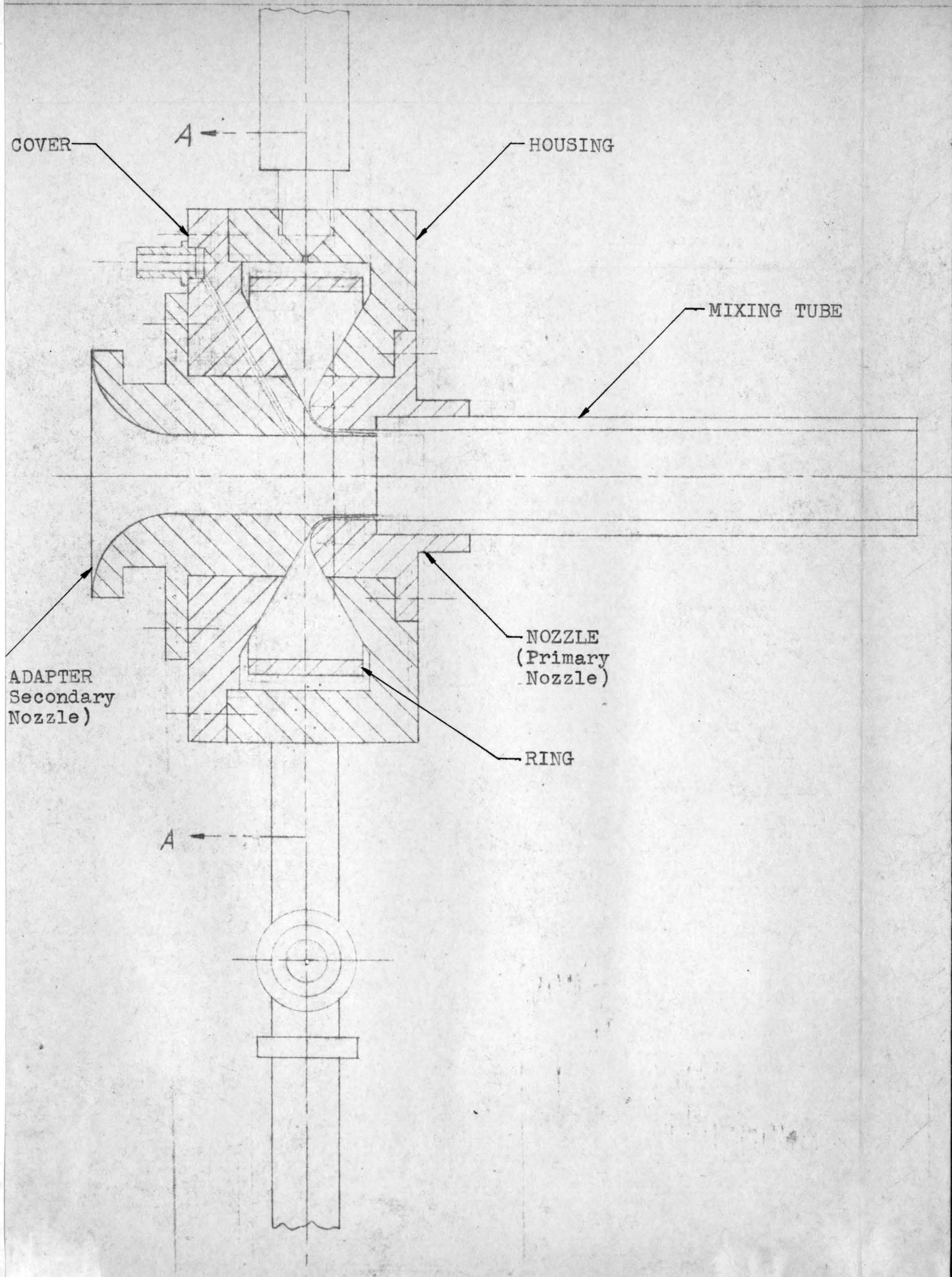
Assuming the ratio of primary pressure to the pressure in the throat of the primary nozzle to be 0.527, the throat velocity can be found from equation 1.

$$V_t = 4.32(589.94)(0.409) = 1035 \text{ feet per sec.}$$

$$\frac{V_1}{V_t} = \left(\frac{P_t}{P_1}\right)^{\frac{1}{K}} = \left(\frac{0.527}{0.155}\right)^{\frac{1}{1.4}} = 2.39$$

$$\frac{A_1}{A_t} = \left(\frac{V_t}{V_1}\right)\left(\frac{V_1}{V_t}\right) = \frac{1035}{1640}(2.39) = 1.51$$

$$A_t = \frac{0.000512(144)}{1.51} = 0.0488 \text{ Sq. ft.}$$



A
COVER
HOUSING
MIXING TUBE
NOZZLE
(Primary
Nozzle)
ADAPTER
Secondary
Nozzle)
RING
A

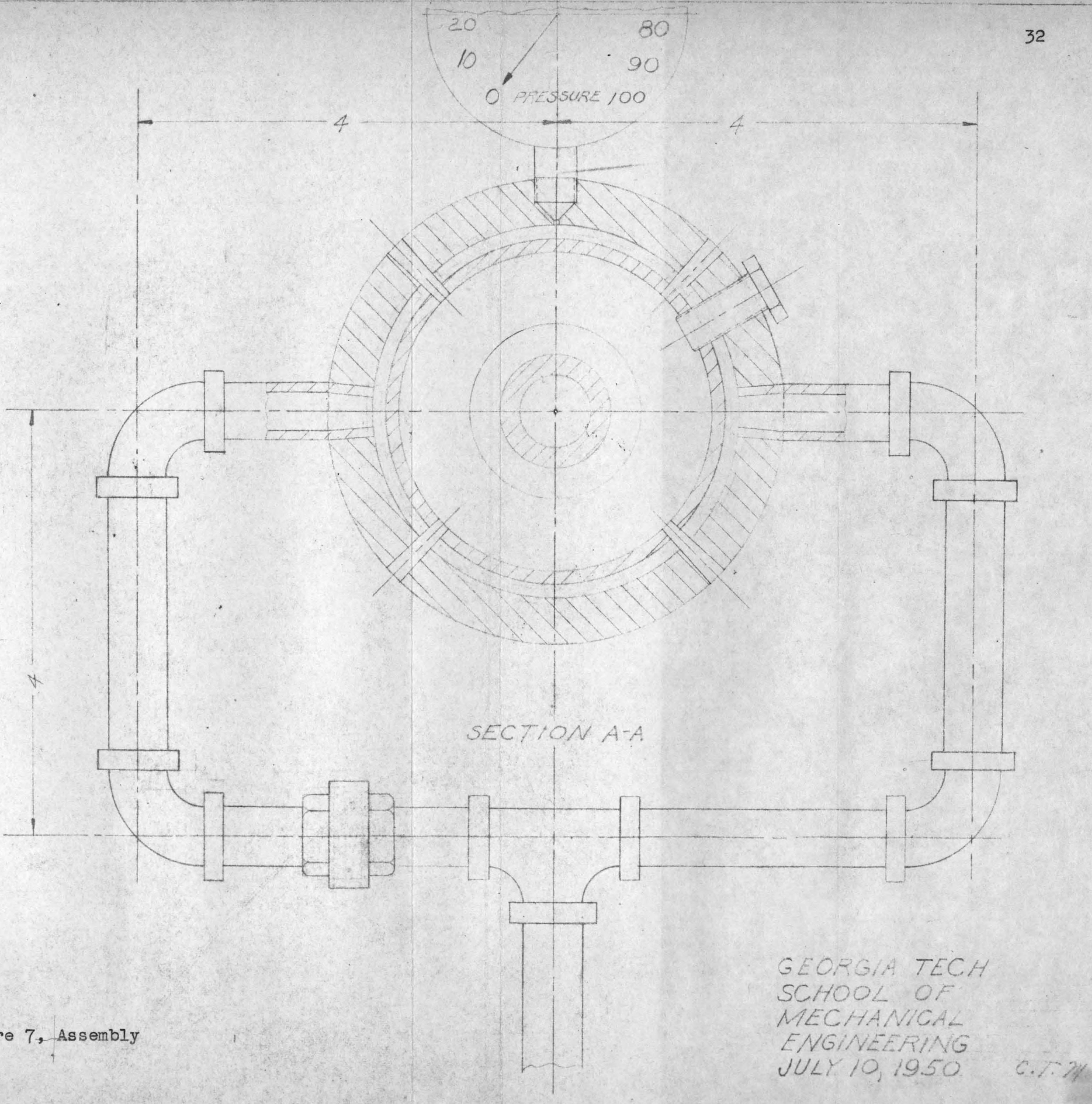


Figure 7, Assembly

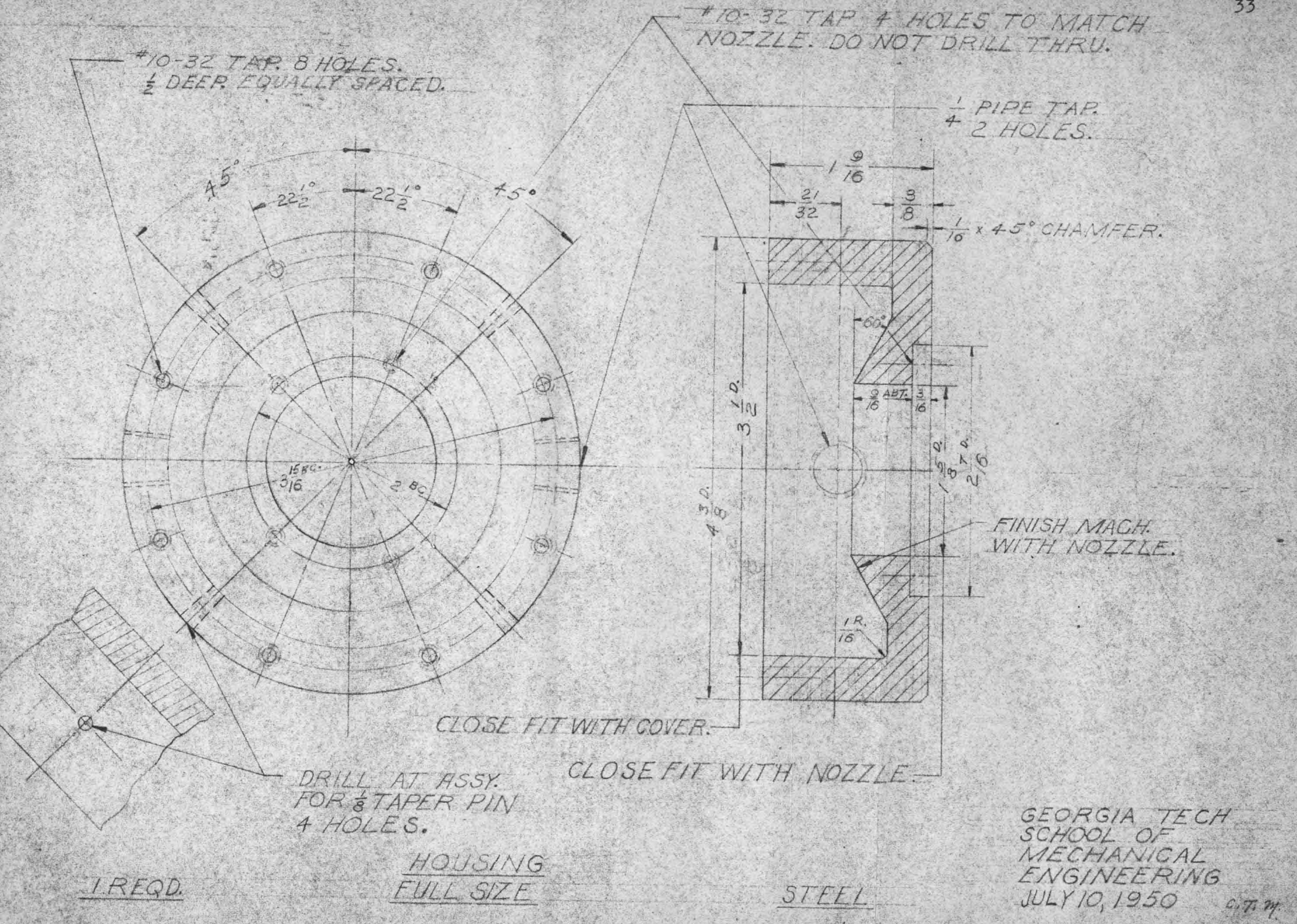

#10-32 TAP 4 HOLES TO MATCH NOZZLE. DO NOT DRILL THRU.
#10-32 TAP. 8 HOLES. 1/2 DEEP. EQUALLY SPACED.
1/4 PIPE TAP. 2 HOLES.
45°
22 1/2°
22 1/2°
45°
1 9/16
21/32
3/8
1/16 x 45° CHAMFER.
60°
9/16 ABT.
3/16
3 1/2 D.
4 3/8 D.
1 5/8 D.
2 7/16 D.
3 15/16 BC.
2 BC.
1/16 R.
FINISH MACH. WITH NOZZLE.
CLOSE FIT WITH COVER.
CLOSE FIT WITH NOZZLE.
DRILL AT ASSY. FOR 1/8 TAPER PIN 4 HOLES.
1 REQD.
HOUSING
FULL SIZE
STEEL
GEORGIA TECH
SCHOOL OF
MECHANICAL
ENGINEERING
JULY 10, 1950
G.T.M.

Figure 8

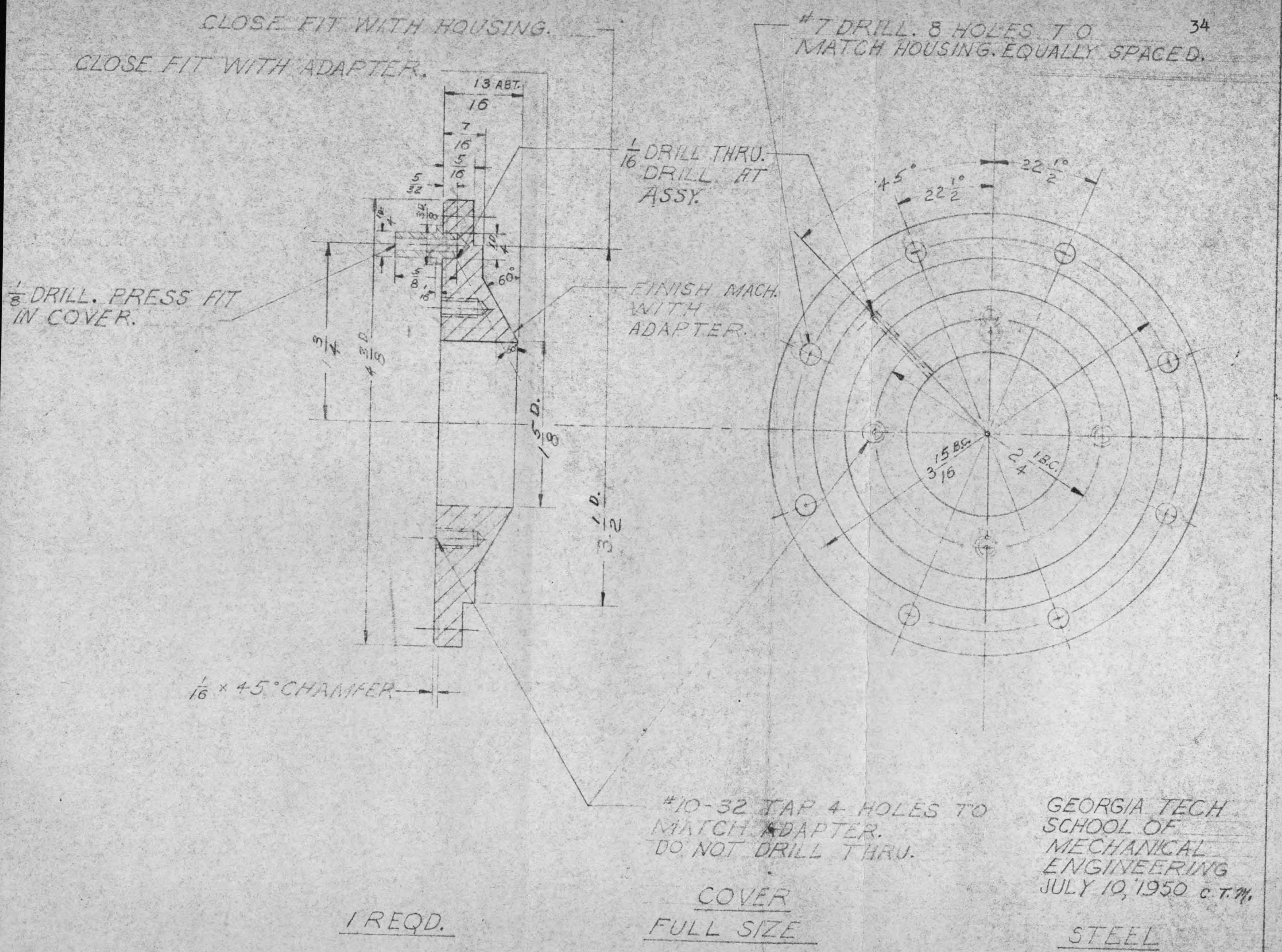

CLOSE FIT WITH HOUSING.
CLOSE FIT WITH ADAPTER.
13/16 ABT.
7/16
5/16
5/32
1/16 DRILL THRU. DRILL AT ASSY.
#7 DRILL. 8 HOLES TO MATCH HOUSING. EQUALLY SPACED.
45°
22 1/2°
22 1/2°
1/8 DRILL. PRESS FIT IN COVER.
60°
FINISH MACH. WITH ADAPTER.
1 3/4
4 3/8 D.
1 5/8 D.
3 1/2 D.
3 15/16 B.C.
2 1/4 B.C.
1/16 × 45° CHAMFER
#10-32 TAP 4 HOLES TO MATCH ADAPTER. DO NOT DRILL THRU.
GEORGIA TECH SCHOOL OF MECHANICAL ENGINEERING JULY 10, 1950 C.T.M.
1 REQD.
COVER
FULL SIZE
STEEL

Figure 9

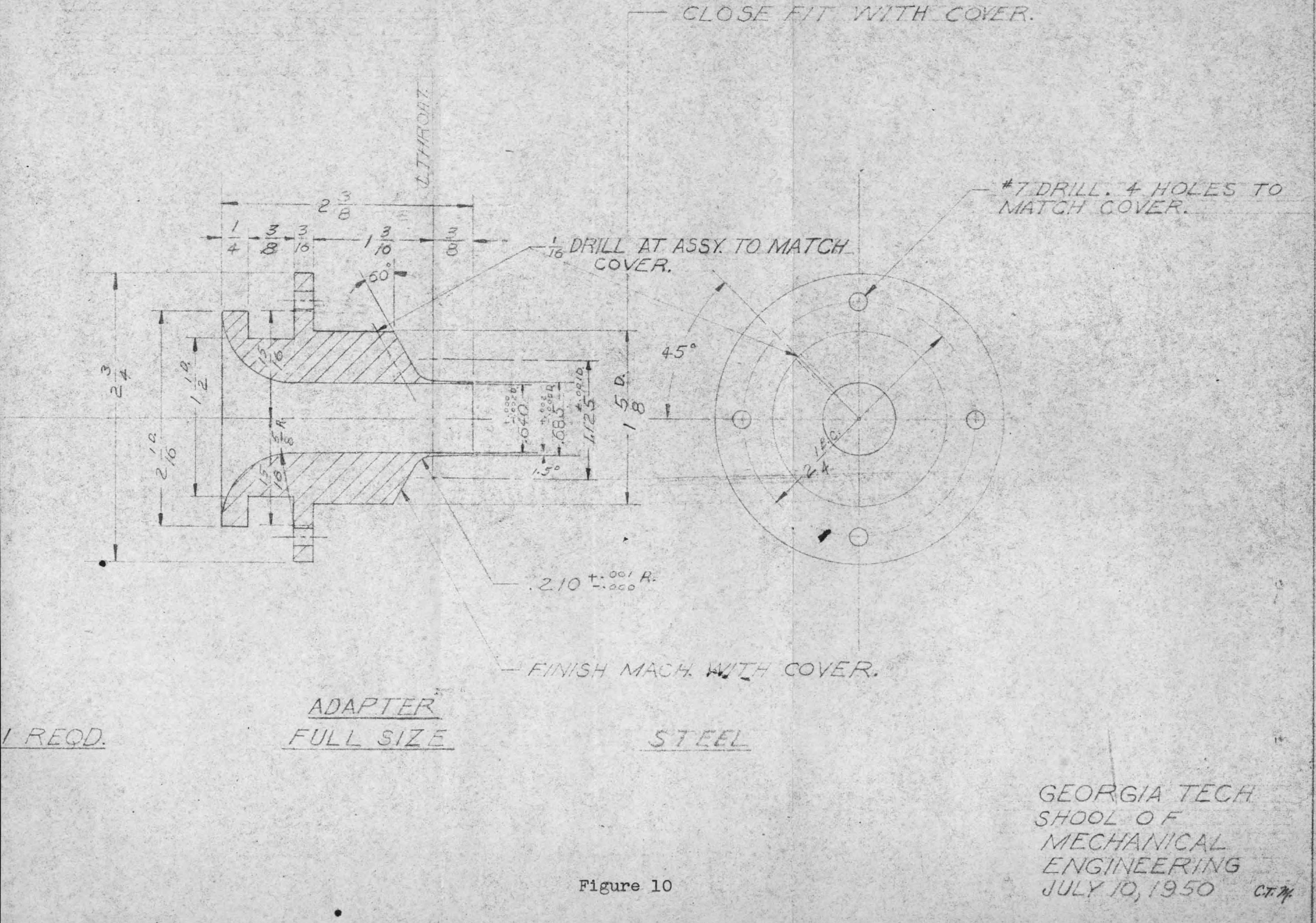
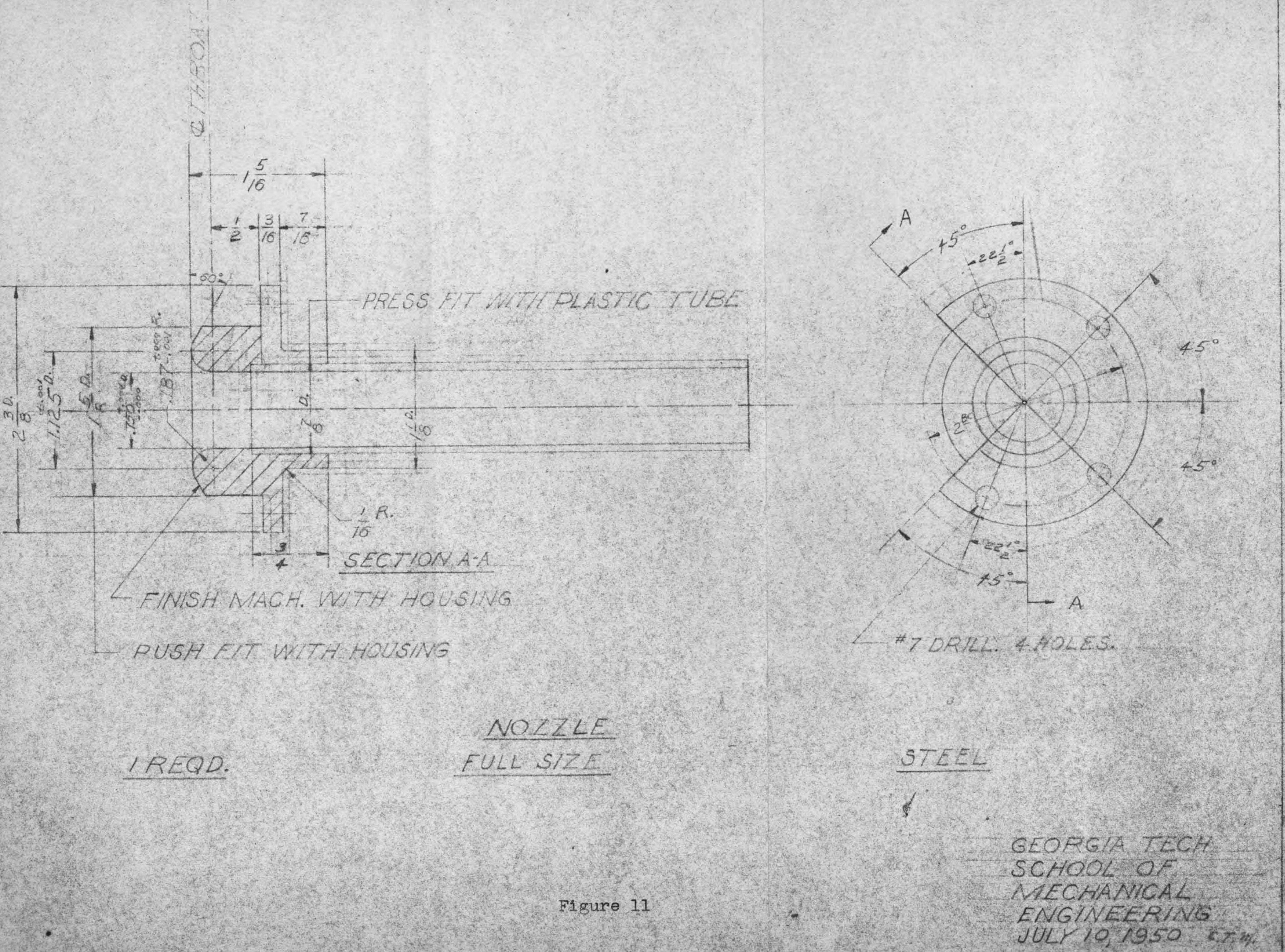
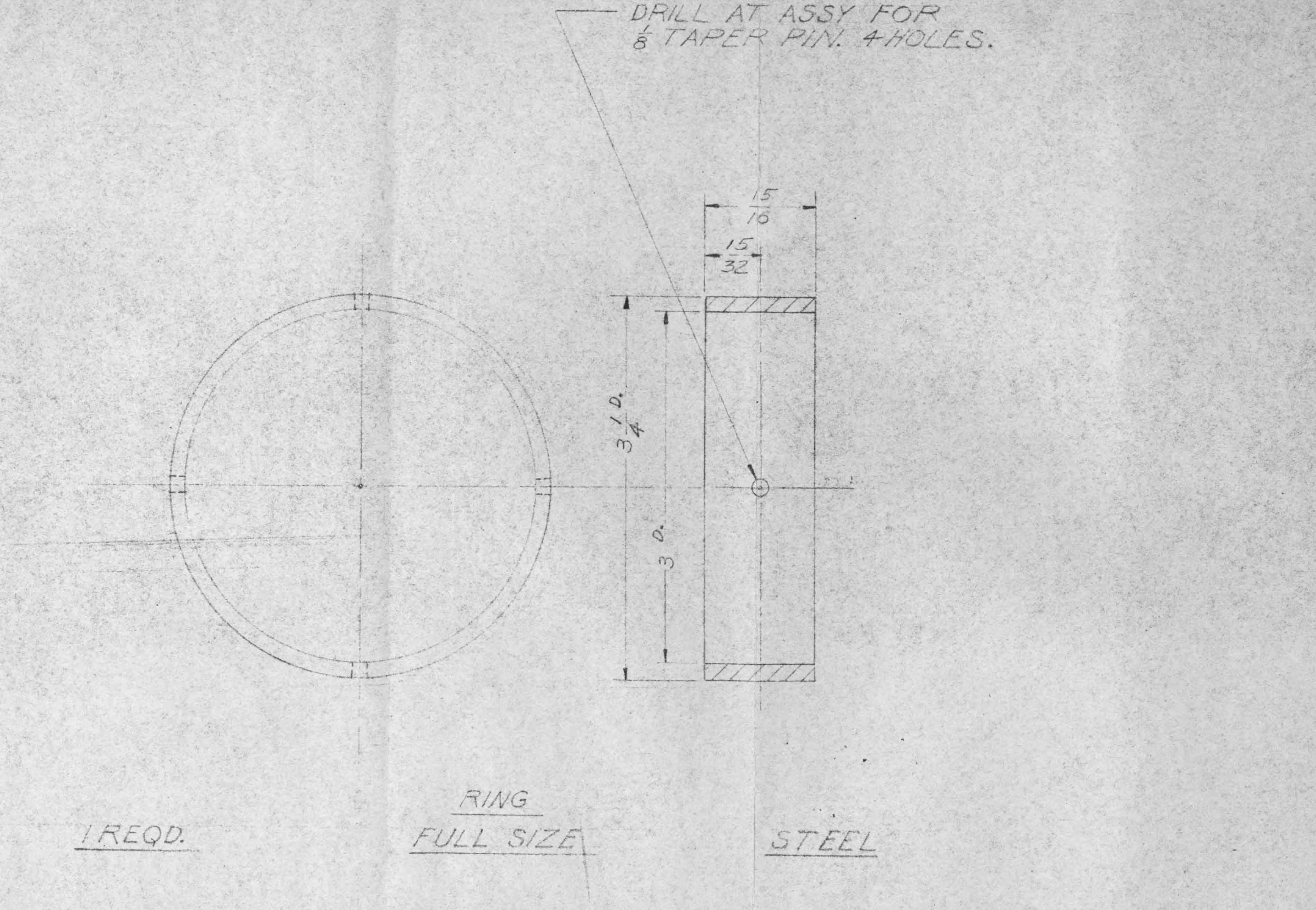

CLOSE FIT WITH COVER.
℄ THROAT
#7 DRILL. 4 HOLES TO MATCH COVER.
1/16 DRILL AT ASSY. TO MATCH COVER.
60°
45°
1 5/8 D.
.210 +.001 -.000 R.
FINISH MACH. WITH COVER.
ADAPTER
FULL SIZE
1 REQD.
STEEL
GEORGIA TECH
SHOOL OF
MECHANICAL
ENGINEERING
JULY 10, 1950
C.T.M.

Figure 10



℄ THROAT
1 5/16
1/2
3/16
7/16
60°
PRESS FIT WITH PLASTIC TUBE
1/16 R.
3/4
SECTION A-A
FINISH MACH. WITH HOUSING
PUSH FIT WITH HOUSING
A
45°
22 1/2°
A
#7 DRILL. 4 HOLES.
1 REQD.
NOZZLE
FULL SIZE
STEEL
GEORGIA TECH
SCHOOL OF
MECHANICAL
ENGINEERING
JULY 10, 1950

Figure 11



GEORGIA TECH
SCHOOL OF
MECHANICAL
ENGINEERING
JULY 10, 1950 C.T.M.

Figure 12